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ABSTRACT

In this paper, variable capacity rotary compressors using By-pass method are investigated. Performance of the instrumented compressor is tested and temperatures at various locations of the compressor are measured. Through the comparison of characteristics between full and part load operations, possibility of capacity control using By-pass method is verified, and some issues at part load operation are clarified. Decrease of motor efficiency, suction gas heating due to the by-passed gas of high temperature, and resistance of gas flow in the by-pass line are main reasons for the decrease in performance at part load operations. For improvement of performance at part load operations, cooling of the by-passed suction gas is carried out and the increment of the performance is proportional to the decrease of suction gas temperature by cooling of the by-passed gas. Variation of the by-passed gas flow rate using needle valve is also carried out, but the effect appears to be very tiny.

INTRODUCTION

Since the air conditioner is running most of times under the range of very low cooling capacity after the initial start-up and the load varies from time to time, the development of variable capacity rotary compressors is one of the very important issues expecially for the purpose of energy saving. As a substitution for previous ON/OFF type compressors, The development of various speed rotary compressors is increasingly active due to the quick response to the change of load resulting in comfortable environment, continuous and wide capacity control and higher efficiency than any other variable capacity compressors [1]. They are, however, not much attractive for the capacity control because of the high initial cost and low reliability at low frequency range. In deed, it is impossible to operate various speed compressors at speeds lower than 30 Hz because the mechanical problems such as the unreliable lubrication occur [2]. On the other hand, the capacity control method by mechanical operation has the cost competitive edge and higher reliability due to the constant speed operation in spite of the relatively low efficiency and discrete control pattern. Furthermore, there is more room for mechanical-type capacity control compressors to be improved in performance because few studies have been done so far on this topic [3, 4, 5].

The objectives of the present study are to observe the heat transfer and performance characteristics of variable capacity rotary compressors using By-pass method and to verify some possibilities to improve their performance at part load operation.

Table 1 Performance characteristics at ASHRAE-T condition

	Full Load	Part Load	Differance
Power Input	1	0.759	24.1%
Capacity	1	0.625	37.5%
E.E.R.	1	0.823	17.7%

EXPERIMENTAL APPARATUS

In this study, the rolling piston type rotary compressor with R-22 was tested and its performance was measured in the compressor calorimeter. Figure 1 shows the schematic of the test compressor fabricated for part load operation test. The lower part including the sub bearing has been modified so that the refrigerant can pass through the pipe that is attached to the by-pass hole on the sub bearing at part load operation. The hole is blocked by a bar so that the compressor is run in normal condition at full load operation. The by-passed refrigerant gas would mix with main flow of refrigerant returning from the system and flow into the accumulator. To measure the temperatures of main portions of compressor and refrigerant, 20 and 7 thermocouples were installed inside and outside the compressor, respectively. Figure 2 shows the positions of temperature measurement and By-pass hole on the cylinder. The By-pass hole is located 180 degree from the vane and the diameter of the pipe is 12 mm. Figure 3 shows the position of the temperature measurement at main portions of the compressor.

EXPERIMENTAL METHODS AND CONDITIONS

Performance characteristics and temperature distributions for full and part load operations were compared at ASHRAE-T condition. Pressure and temperature of the by-passed gas at the exit of the cylinder were additionally measured at the part load operation. In order to observe the performance characteristics of the compressor at the various operation conditions, the experiments were conducted at three different evaporation and condensation temperatures for the part load operation and their measurements were compared with those for full load operation. The by-passed refrigerant that was already heated in the cylinder returns to the compressor and increases the temperature of the suction gas, which results in the reduced performance of compressor. Therefore, the by-passed refrigerant was cooled by external fan to observed the variation of temperatures and performance with the decrease of temperature of the by-passed gas. Furthermore, possibility of capacity control in a tiny range with fixed by-pass structure was tested with a needle valve. Figure 4 shows the schematic of experimental apparatus for cooling and mass flow control tests.

EXPERIMENTAL RESULTS AND DISCUSSIONS

1. ASHRAE-T condition

1.1 Comparison of performance

Table 1 compares compressor performances of full load and part load operation conditions at ASHRAE-T condition. The capacity at part load operation was 62.5% of that at full load operation condition when the by-pass hole is located at 180 from the position of vane. This value is higher than what was expected from the geometric point of view. The pressure measured at (A) of Fig. 4 is 0.538 MPa (5.49 kg_f/cm^2), which is a little bit higher than suction pressure, indicating that the refrigerant is resisted in the by-pass pipe and slightly compressed. Lida et al. [3] shows that the cooling capacity is over 70% when the by-pass hole locates at the same position. The input power is 75.9% of that at the full load condition so that EER at part load operation goes down to 82.3% in result. The main reasons that the reduction of input power is smaller than that of capacity are due to the constant speed motor regardless of the load and the decrease of motor efficiency at part load operation. The reduction of the input power is believed to be mainly due to the reduction of compression work for By-pass capacity control method, because constant-speed operation consumes same amount of power.

Figure 5 indicates the characteristic curves of typical motors used for rotary compressors. Point (A) and (B) represent operation points for full load and part load, respectively. The point (B) can be obtained by applying measured load at the part load to the torque curve. When the compressor was run at the part load, the rpm increases 1.1% while the motor efficiency decreases about 2.8% compared with values for full load operation. Even if the effect of change of rpm on the cooling capacity and input power is analyzed to be small, the 2.8% reduction of motor efficiency would reduce the compressor performance seriously since the motor loss accounts for the largest portion among compressor losses [6]. If the operating point of compressor is adjusted to fit the part load operation, the SEER (seasonal energy efficiency ratio) can be improved despite of the small reduction of performance at the full load point.

1.2 Comparison of temperature distributions

Figure 6 shows the temperature distributions for part load and full load operations at ASHRAE-T condition. The overall trend is similar while the part load operation shows higher temperatures at suction part and cylinder than those for full load operation. The temperature of the refrigerant that was by-passed from the compressor is 75.3℃ and the pressure is 0.538 MPa (5.49 kg_f/cm^2) at point A of Fig. 4, which is little bit higher than suction pressure. The by-pass refrigerant is presumed to flow uncompressed out of the compressor since the temperature of the cylinder does not change form point 2 to 3 in Fig. 2 and the pressure at the by-pass line is not increased a lot from the suction pressure. Therefore, it is believed that the re-expansion of the by-passed gas to the cylinder is not significant. The temperature of the by-passed refrigerant goes down to 69.3℃ at point B of Fig. 4 by heat transfer with environment along the circulation pipe. The temperature of refrigerant coming to the accumulator, which is theoretically calculated, turns out to be 47.9℃ and 12.9℃ higher than the suction gas temperature at full load operation. It is known that high suction temperature increases the specific volume of the refrigerant so that not only the cooling capacity decreases but also the increased overall temperature of compressor causes some problems on the compressor. In this study, it is concluded that the volumetric efficiency has been reduced a lot because the reduction of the cooling capacity is bigger than the decrease of the input power. However, it is also observed that the overall temperature distribution did not change much even at the part load operation probably because the exit condition of the compressor in the calorimeter was fixed and the reduced amount of refrigerant under compression compensated the increase of the temperature.

Figure 7 compares the temperature distributions on main parts around the motor at full and part load operations. When the motor is run at part load condition, the overall temperature of motor section decreases even if the difference is less than 1℃. It is because the reduced input power at part load operation lowers the heat loss compared with full load operation and results in the reduction of heat dissipation from the motor. The heat dissipation from the motor at part load condition, which could be calculated from Fig. 5, was decreased by 13% compared with that at full load operation. The temperature drop seems to be even bigger than what is shown in Fig 7 if the heat transfer from the cylinder, which is relatively at higher temperature, is taken into account. In Fig. 7, the temperature of the refrigerant passing through the gap between rotor and stator of the motor decreases because the heat was transferred to the motor at part load operation, while the temperature increases at full load condition because the heat was transferred from the motor. It also indicates that the temperature of the motor part was definitely decreased at part load operation.

2. Variation of evaporation and condensation temperatures

Figure 8 shows the change of EER with respect to the variation of evaporation and condensation temperatures at part load operation. The variation of performance characteristics with the change of operation condition at part load is about the same as that at full load operation. In other words, since the rate of variation of EER is the same for part and full load operations, it is concluded that the performance with the by-pass type capacity control technique does not depend on the operation conditions such as evaporation or condensation temperatures. The change of temperature distribution at the part load operation (which is not shown in this paper) is also in accord with that at the full load operation.

3. Cooling of by-passed refrigerant

It is known that one of the most serious problem with part load operation is the decrease of volumetric efficiency due to the increase of the suction gas temperature. Therefore, the relationship between the compressor performance and the suction gas temperature with the cooling of the by-passed refrigerant was investigated. Figure 9 shows the improvement of the performance with the reduction of temperature of by-passed suction gas by cooling. As the increase in the performance is linearly proportional to the reduced temperature at the part load operation, EER increases by 3.5% compared with that at AHSRAE-T condition when the suction gas temperature is reduced by 23℃. This value represents 85.2% of EER at the full load operation. Since the EER value for part load operation without cooling is 82.3% of that at the full load operation, cooling of by-pass refrigerant results in the improvement of performance by 2.9%. The input power of the compressor is observed to be constant regardless of cooling. Because the mass flow rate of the refrigerant is increasing while the compression work per unit mass decreases due to the reduction of specific volume of refrigerant by cooling, the input power for the compressor ends up constant. Figure 10, which shows the variation of temperature distribution with cooling of by-passed suction gas, indicates that the overall temperature decreases as much as the reduction of the suction gas temperature.

4. Control of by-passed gas flow rate

In order to investigate the possibility of controlling the flow rate of by-passed refrigerant in a small range, the opening of needle valve installed in the by-pass line was controlled from point A to C as shown in Fig. 11. Figure 11 shows the change of cooling capacity with respect to the value opening. It was presumed before the experiments that the cooling capacity would be reduced due to the increase in amount of by-passed refrigerant as the valve opening is increasing. However, it was turned out that the cooling capacity increased and the amount of change was too weak to apply. It was, therefore, concluded that needle value was not able to control the capacity when the position of by-pass hole and the size of the pipe are fixed. The variations of the pressure at point A and the temperature at point B with respect to the valve opening are given in Fig. 12. Since the pressure at A increases as the valve is closing, the effect of re-expansion of refrigerant is believed to be increased due to the increased pressure inside the cylinder. The temperature distribution (not shown in this paper) shows that the overall temperatures of the compressor including the cylinder increase while the suction gas temperature entering the compressor through the needle valve decreases, which indicating that the effect of re-expansion of refrigerant is increasing.

CONCLUSIONS

In the present study, the possibility of capacity control of rotary compressors by By-pass method, and the performance and heat transfer characteristics at part load operation were investigated. To improve the performance of the compressor at the part load condition, tests such as the cooling of the by-passed refrigerant and the flow rate control with needle vale were conducted.

It was confirmed that the by-pass method is able to control the capacity of rotary type compressor and reduce the input power at the part load operation. The motor efficiency decreases by 2.8% at part load operation, which makes a great contribution to the reduction of compressor performance. The temperature of suction gas increases greatly by the by-passed gas and, therefore, the performance of the compressor can be increased by 2.9% compared to that at full load operation by cooling the by-passed refrigerant. The performance characteristics with variation of operation condition at part load operation are very similar to those at full load operation. It was also observed that the flow rate control with a needle valve in the by-pass pipe was not useful to control the capacity of the compressor.

ACKNOWLEDGMENTS

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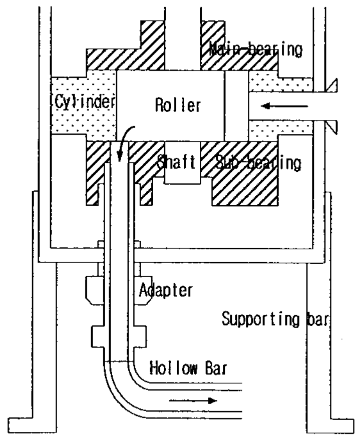


Fig. 1 Schematic of test compressor at part load operation

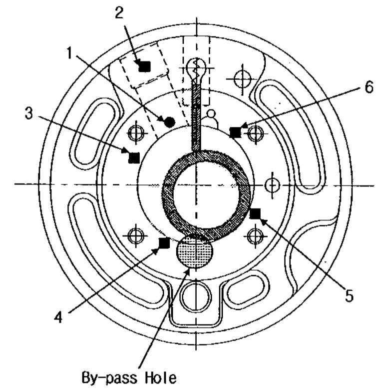


Fig. 2 Positions of temperature measurement and By-pass hole on cylinder

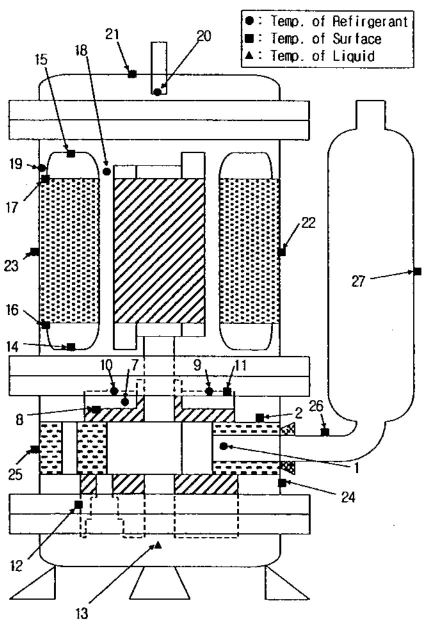


Fig. 3 Positions of temperature measurement

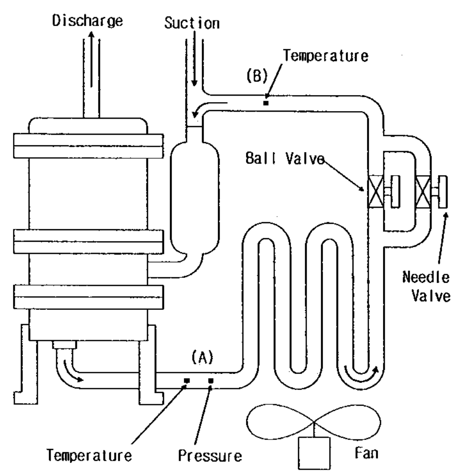


Fig. 4 Experimental apparatus for cooling and mass flow control test

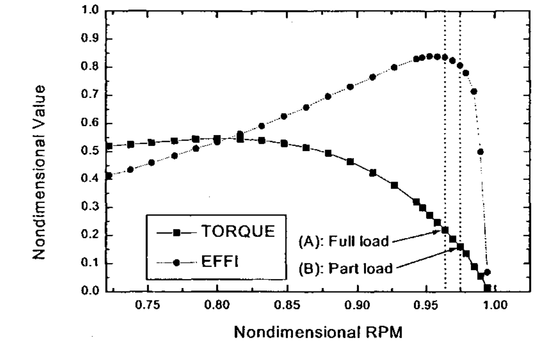


Fig. 5 Characteristics of motors for rotary compressor

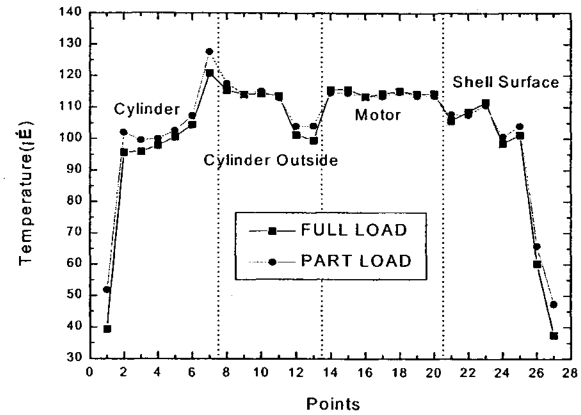


Fig. 6 Comparison of temperature distributions at part and full load operations

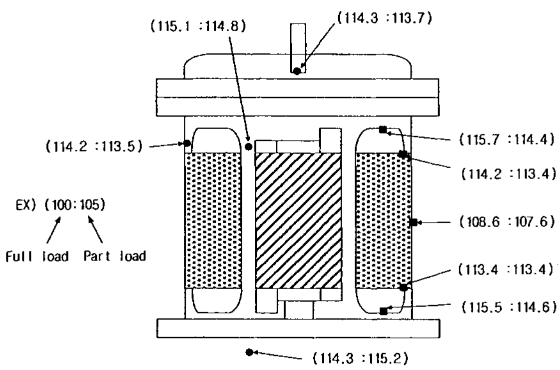


Fig. 7 Comparison of temperatures at upper part of compressor

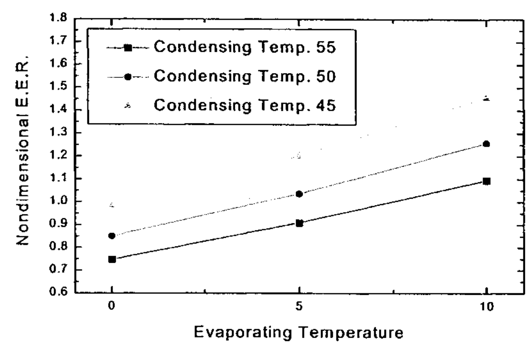


Fig. 8 Variation of nondimensional EER with variation of eva. and cond. temperatures

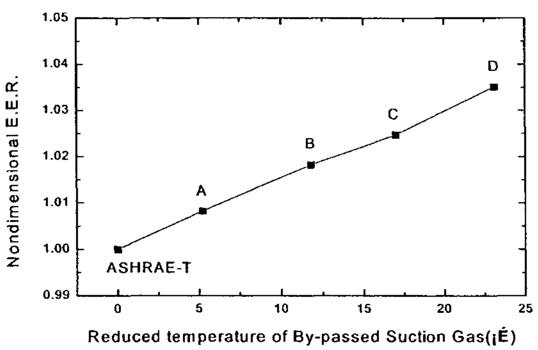


Fig. 9 Variation of nondimensional EER with cooling of by-passed suction gas

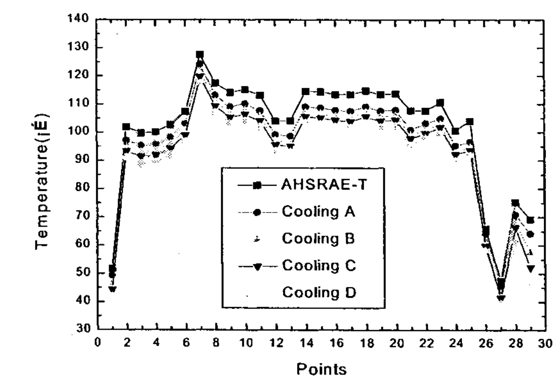
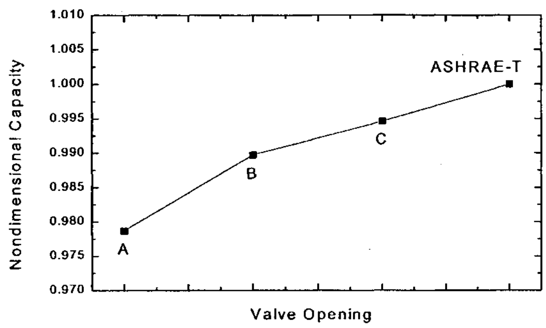


Fig. 10 Variation of temperature with cooling of by-passed suction gas



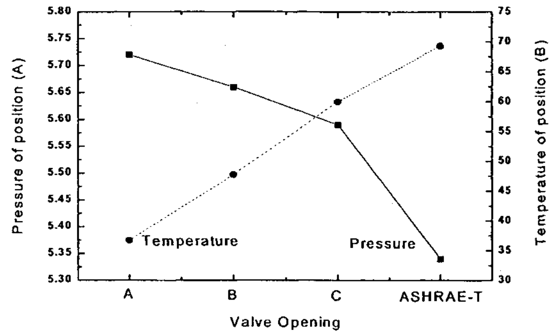


Fig 11 Variation of nondimensional capacity with valve opening

Fig 12 Variation of pressure of position A and and temperature of position B with valve opening